

Research Article

CFD and Experimental Study of Cavitation Behavior in Three-Lobe Journal Bearings Lubricated with TiO_2 -Based Nanofluids

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Abstract:

Three-lobe journal bearings are widely used in high-speed rotating machinery due to their superior stability and load-carrying capacity. However, they are susceptible to vapor cavitation in the diverging region, which weakens the lubricant film and degrades performance. This study presents a combined numerical and experimental investigation of pressure distribution in three-lobe journal bearings under cavitating conditions, comparing a base oil (Mobil DTE 24) with a TiO_2 -enhanced nanolubricant. Simulations were performed using the Zwart–Gerber–Belamri (ZGB) cavitation model within a multiphase CFD framework in ANSYS Workbench 2024R1 at operating speeds of 500–1000 rpm. Results were validated against experimental pressure measurements at twelve angular positions using a custom-built test rig. For the base lubricant, CFD accurately predicted peak pressure near 210° and vapor cavitation between 240° and 330° , with deviations within ± 0.06 MPa. The addition of TiO_2 nanoparticles increased peak pressure by up to 36% and raised the minimum pressure from -0.52 MPa to $+0.50$ MPa, indicating over 60% cavitation suppression. These improvements confirm enhanced film stability, greater load capacity, and close agreement between simulation and experiment. Furthermore, the TiO_2 nanolubricant reduced friction losses, minimized lubricant degradation, and supports the development of energy-efficient and sustainable tribological systems.

Keywords: Three-lobe journal bearing, TiO_2 nanolubricants, CFD, Cavitation effect, Hydrodynamic lubrication

1. Introduction

Hydrodynamic journal bearings (HJBs) are critical components in rotating machinery because of their high load-carrying capacity, reliability, and smooth operation. However, conventional cylindrical bearings, though structurally simple, tend to exhibit instability at high speeds, leading to oil whirl and frictional power losses [1,2]. To overcome these issues, alternative geometries such as two-lobe, three-lobe, and four-lobe configurations have been introduced. Among these, three-lobe bearings are particularly effective due to their inherent preload and segmented pressure zones, which enhance film stiffness and dynamic stability [3–5].

Advancements in computational fluid dynamics (CFD) have enabled detailed modeling of multiphase and cavitating flows in bearings. Gertzos et al. analyzed three-dimensional Bingham lubricant flows [6], while other studies employed the mixture model combined with the Zwart–Gerber–Belamri (ZGB) cavitation model to capture vapor formation in expanding regions [7,8]. Dhande et al. incorporated fluid–structure interaction (FSI) and achieved results

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within a 5% deviation from experimental data [4]. The ZGB model has since become a standard approach for cavitation modeling, accurately representing bubble generation and collapse in hydrodynamic systems [9–11].

In parallel, research on lubricant rheology and nano-enhancements has advanced bearing performance. Studies have highlighted the influence of non-Newtonian lubricant behavior on pressure generation [12,13] and proposed multi-objective optimization strategies for minimizing friction losses while improving film thickness and stability [14]. From a surface engineering standpoint, texturing techniques such as micro-protrusions and dimples have enhanced load capacity and stiffness, though sometimes at the expense of dynamic margin [3,5,15].

Cavitation remains one of the most critical issues affecting bearing durability and efficiency. Experimental evidence confirms that cavitation erosion reduces bearing life and load capacity [11,16]. CFD analyses of elliptical journal bearings using TiO₂ and ZnO nanofluids have shown that nanoparticles enhance viscosity and reduce side leakage under thermal effects [17]. TiO₂ nanolubricants, in particular, possess excellent tribological and thermal properties, reducing friction and wear while maintaining stable viscosity [18]. Multi-lobe bearings lubricated with nanofluids have demonstrated improved load-carrying capacity and reduced friction, especially at moderate speeds ($N < 1000$ rpm), with minimal temperature rise (<10 °C) [16,19–21]. Incorporating TiO₂ nanoparticles into base oils enhances thermal management and viscosity retention [22,23], while preload and eccentricity further influence load capacity and leakage in three-lobe bearings [24].

Recent studies report that TiO₂ nanofluids at concentrations up to 2 wt.% can increase load capacity by more than 14% without a significant temperature rise [25]. Optimized particle size and concentration improve viscosity, pressure development, and film stability [26]. Other investigations confirm that TiO₂ nanolubricants enhance pressure, mitigate cavitation, and improve lubrication performance across various operating conditions [27–29]. However, most previous studies have either overlooked cavitation modeling or lacked experimental validation, particularly for complex geometries such as three-lobe bearings.

This study fills a research gap by presenting the first combined CFD-experimental evaluation of TiO₂ nanofluids in three-lobe bearings under cavitation conditions. The mixture model integrated with the ZGB cavitation framework is used and validated through experiments conducted on a custom-built test rig operating at 500, 750, and 1000 rpm under a constant 1 kN radial load. Pressure was measured at twelve angular positions using high-precision sensors. The results show up to a 36% increase in peak pressure and approximately 60% reduction in cavitation with TiO₂ nanolubricants, along with excellent agreement (within $\pm 6\%$) between CFD and experimental data. These findings confirm the effectiveness of TiO₂ nanofluids in enhancing film stability and hydrodynamic performance, establishing a foundation for future applications in smart and energy-efficient bearing systems.

2. Methodology

The study integrates CFD simulation and experimental validation to analyze pressure distribution and cavitation in a three-lobe journal bearing operating at 500–1000 RPM under a constant 1 kN load and an eccentricity ratio of 0.6. A structured hexahedral mesh and the mixture model with the ZGB cavitation approach are implemented in ANSYS Workbench 2024R1. Lubrication using 0.5% wt. TiO₂ nanofluid (Mobil DTE 24 base oil) is modeled and validated through a custom-built test rig with 12 pressure ports. In the present analysis, the TiO₂-blended lubricant was treated as a Newtonian fluid, as the nanoparticle concentration (0.5 wt.%) is relatively low and does not cause noticeable shear-thinning or viscoelastic effects. Previous studies have reported that TiO₂ nanolubricants below 1 wt.% concentration exhibit nearly constant viscosity under moderate shear rates, validating the Newtonian assumption [12, 13, 17]. Moreover, the bearing operates under isothermal and laminar flow conditions, where shear rate variation is minimal. Therefore, modeling the nanofluid as Newtonian with pressure- and temperature-dependent viscosity (as per Singla et al., 2014 [30]) provides sufficient accuracy without introducing unnecessary model complexity.

2.1 Bearing geometry

This study investigates pressure distribution and cavitation in a three-lobe journal bearing operating at 500, 750, and 1000 RPM under a constant 1 kN load. The three-lobe journal bearing, shown in Figure 1, features a preload factor of 0.5, 20° axial grooves, and a radial clearance of 0.055 mm, promoting a converging-diverging film profile essential for pressure development.

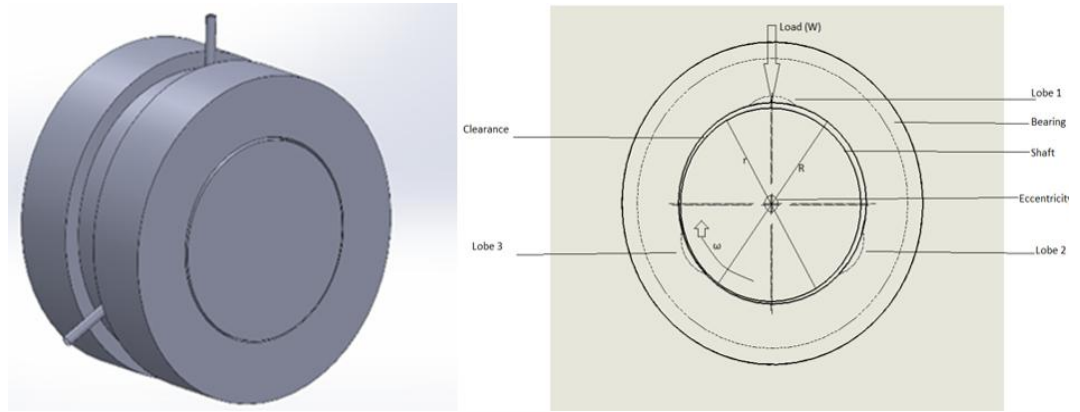


Fig. 1. Three-lobe bearing geometry.

All essential input parameters used in the simulation and experiments including bearing geometry, journal and bearing materials, lubricant type, nanoparticle concentration, and thermophysical properties are summarized in Table 1. These values are based on experimental measurements, material specifications, and literature data.

Table 1: Parameters used in the analysis.

Parameter	Symbol	Value	Unit
Journal diameter	d	49.89	mm
Bearing diameter	D	50	mm
Radial clearance	C	0.055	mm
Preload factor	δ	0.5	-
Bearing length	L	50	mm
L/D Ratio	-	1	-
Rotational speed	N	500, 750, 1000	RPM
Applied load	W	1000	N
Lubricant	-	MOBIL DTE 24 with 0.5% wt.TiO ₂ nanoparticles	-
Lubricant dynamic viscosity (@40°C)	μ	0.0292 (33.77 cSt)	Pa·s
Inlet Pressure	P	0.4	MPa
Eccentricity ratio	ε	0.6	-
Lubricant density	ρ	869	kg/m ³
Vapor saturation pressure	P_v	0.018542	MPa
Vapor dynamic viscosity	μ_v	2×10^{-5}	Pa·s
Bearing material	-	C45 Steel: $E = 200$ GPa, $\rho = 7870$ kg/m ³ , $\nu = 0.3$	-
Shaft material	-	EN8 Steel: $E = 210$ GPa, $\rho = 7850$ kg/m ³ ,	-
Nanoparticle type	-	Titanium dioxide (TiO ₂), 0.5% wt.	-
TiO ₂ nanoparticle size	-	40 nm (Range: 30–50 nm), $\rho = 4.23$ g/cm ³	-
Specific heat (lubricant)	C_p	1951	J/kg·K

2.2 Computational mesh generation

The fluid domain was discretized with a structured hexahedral mesh in ANSYS Workbench 2024R1, as shown in Figure 2. To accurately resolve the viscous sublayer near the journal and bearing surfaces, seven inflation layers were applied with a first-layer thickness of 1 μ m. Particular emphasis was placed on preserving mesh integrity in critical hydrodynamic regions, especially the pressure development zone, where skewness was maintained below 0.3. The overall mesh quality remained within acceptable limits, with skewness values ranging from 0.16 to 0.85.

A mesh independence study was conducted to ensure numerical reliability, revealing less than 2% variation in key output parameters with further refinement. This confirmed that the adopted mesh configuration was sufficient for capturing pressure distribution and cavitation behavior without incurring excessive computational cost.

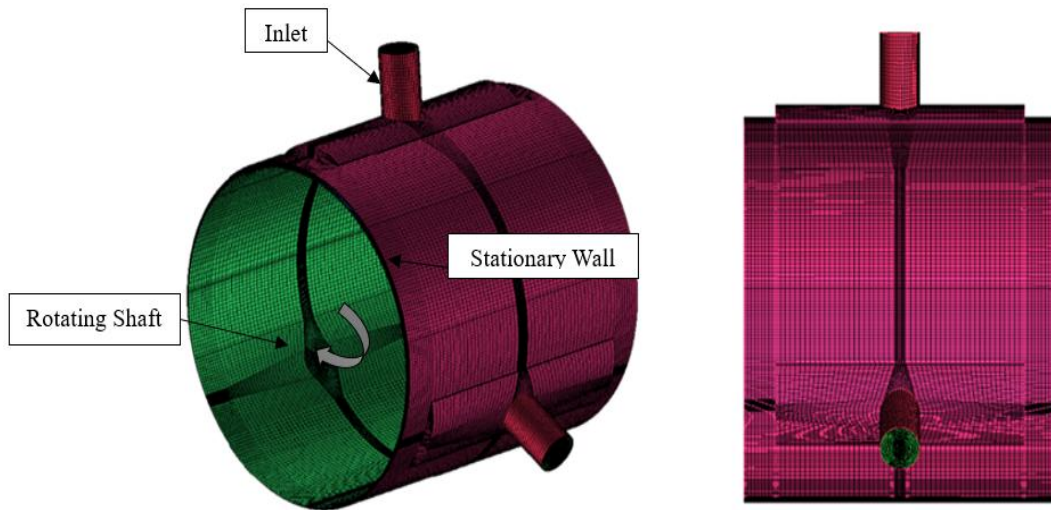


Fig. 2. Structured hexahedral mesh with 7 inflation layers.

The CFD analysis was performed using ANSYS Workbench 2024R1 with a pressure-based, 3D, double-precision, steady-state solver under a laminar flow regime. Cavitation effects were modeled using the Zwart–Gerber–Belamri (ZGB) model within a mixture multiphase framework incorporating a scalar vapor transport equation. The inlet was defined as a pressure inlet with a gauge pressure of 0.4 MPa at 0°, while the outlet was set to atmospheric pressure. The journal wall was modeled as a rotating surface at 500, 750, and 1000 RPM, and the bearing wall was stationary with a no-slip condition. A structured hexahedral mesh consisting of 415,716 cells with seven inflation layers was used to ensure near-wall resolution. The mesh quality was satisfactory, with skewness values between 0.16 and 0.85 and critical regions below 0.3. A high-resolution advection scheme was applied, and convergence was achieved when residuals for continuity and momentum equations fell below 10^{-6} .

To capture this multiphase behavior numerically, various cavitation-capable CFD models were evaluated. The Volume of fluid (VOF) method, while effective in free-surface flow simulations, lacks resolution for sub-millimeter cavitation structures. The discrete phase model (DPM) is restricted to dilute vapor regimes, making it unsuitable for high-density cavitating flows such as those in journal bearings. The Eulerian model, though highly accurate, is computationally intensive and less practical for large-scale parametric studies. The mixture model was chosen due to its optimal balance between computational efficiency and predictive accuracy. This model solves shared mass and momentum equations while tracking the vapor phase using a scalar transport equation, which effectively captures vapor fraction variations within the lubricant film [7,8].

Cavitation was modeled using the **Zwart–Gerber–Belamri (ZGB)** model, which simplifies the **Rayleigh–Plesset** bubble dynamics with <5% deviation for submicron bubbles typical in journal bearings. The detailed derivation of the model is well established in previous studies [4,9,11]; hence, only the key governing relations are presented here. The resulting bubble growth rate is expressed in Eq. (1):

$$\frac{dR_b}{dt} = \sqrt{\frac{2}{3} \cdot \frac{p_b - p}{\rho_l}} \quad (1)$$

This relationship is embedded into the vapor transport Eq. (2):

$$\frac{\partial}{\partial t}(a_v \cdot \rho_v) + \nabla(a_v \cdot \rho_v \cdot v_v) = C_e - C_c \quad (2)$$

Where, C_e and C_c account for the mass transfer between liquid and vapour phases in cavitation and a_v is the vapor volume fraction and is key to accurately representing local density variations in cavitated zones.

2.3 Governing equations

The hydrodynamic behavior within the journal bearing is governed by the Reynolds equation, which is derived from the conservation laws of mass and momentum considered [4]. These equations were numerically solved using ANSYS Workbench 2024R1. The mass conservation equation is expressed in Eq. (3):

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \cdot \vec{v}) = 0 \quad (3)$$

Where, ρ is fluid density and $\vec{v}$ is the fluid velocity vector.

The momentum Eq. (4) is as follows:

$$\frac{\partial(\rho \cdot \vec{v})}{\partial t} + \nabla(\rho \cdot \vec{v} \cdot \vec{v}) = -\nabla P + \nabla(\vec{\tau}) + \rho \cdot \vec{g} + \vec{F} \quad (4)$$

Where P is the static pressure, $\vec{\tau}$ is the stress tensor, $\rho \cdot \vec{g}$ is the force due to gravity, and $\vec{F}$ is the external body force.

Flow regimes were characterized using Reynolds number (R_e), calculated using Eq. (5):

$$R_e = \frac{\rho \cdot \omega \cdot R \cdot C_b}{\mu} \quad (5)$$

Values between 4.28 and 8.56 across the tested speeds confirm laminar flow, well below the turbulent threshold ($R_e < 2000$).

The present analysis is based on the hydrodynamic lubrication (HL) regime. Although Eqs. (6) and (7) include deformation and viscosity terms for completeness, the simulation was performed under isothermal and non-deforming conditions, as elastic deformation is negligible in C45 steel ($E = 200$ GPa) at the applied load of 1 kN and speed range of 500–1000 RPM. Similar assumptions have been adopted in previous studies [4, 16, 21].

To capture film thickness variation with journal motion and preload geometry, the film thickness expressed as shown in Eq. (6) [4].

$$h = c_p - X \cdot \cos(\theta) - Y \cdot \sin(\theta) - (c_p - c_b) \cdot \cos(\theta - \theta_p) + \delta_E \quad (6)$$

Where, h is the film thickness, c_p is the preloaded radial clearance; X and Y are journal center displacements; θ is the angular position; c_b is the base circle clearance; θ_p is the preload (lobe) angle; and δ_E is elastic deformation.

The lubricant viscosity was modeled as a function of both pressure and temperature, using the expression from Singla et al. (2014), as shown in Eq. (7):

$$\mu = \mu_0 e^{\alpha(P-P_0)} e^{\beta(T-T_0)} \quad (7)$$

Where α and β are empirical pressure and temperature coefficients. Due to the minimal temperature rise (<10 °C) reported at these operational speeds, full thermo-structural coupling was excluded [30].

2.4 Experimental setup

The journal bearing test rig, as illustrated in Figure 3, consists of a base frame, journal assembly, belt-driven A.C. motor, loading mechanism and power pack. The setup accommodates both circular and non-circular bearings. Lubrication is provided by a hydraulic system using Mobil DTE 24 oil, circulated at 0.4 MPa via a 0.5 hp gear pump. Pressure measurements are taken using a Keller Series 35X transducer (-1.5 to $+3$ MPa, $\pm 0.1\%$ FS accuracy), and temperatures are monitored with K-type thermocouples (0 – 600 °C). The bearing includes 12 circumferential sensor ports at 30° intervals for comprehensive pressure profiling. The system runs for 30 minutes to reach steady-state before data collection.



Fig. 3. (a) The power pack unit; (b) The arrangement for the experiment; (c) The setup for pressure sensors, along with load and bearing installation; (d) The test bearing equipped with twelve evenly spaced openings for mounting pressure sensors.

After powering the system and verifying oil circulation, the journal is rotated using the Variable frequency drive (VFD), and the load is applied with a spring balance. Once steady conditions are achieved, pressure readings are recorded at all 12 positions.

3. Results and discussions

The incorporation of TiO_2 nanoparticles into journal bearing lubrication markedly improves pressure stability, suppresses cavitation, and enhances overall hydrodynamic performance. Improved film formation and thermal characteristics contribute to higher load-carrying capacity and operational reliability. The results show strong agreement between CFD predictions and experimental observations.

3.1 CFD and experimental results

The effect of cavitation on pressure distribution was numerically evaluated using the ZGB model under varying shaft speeds, comparing base oil with nanolubricant. As illustrated in Figure 4(a–c) and detailed in Table 3, the nanolubricant consistently increased peak film pressures and reduced negative pressure zones across all speeds. At 500 RPM, cavitation was visibly suppressed in the diverging zone, while at 1000 RPM, near-complete elimination of vapor regions was achieved with the nanofluid. These results validate the enhanced pressure generation and cavitation resistance offered by TiO_2 additives, contributing to improved film stability and load-carrying performance. In each subfigure, the left plot corresponds to the base oil (without nanoparticles), and the right plot shows the TiO_2 -enhanced lubricant.

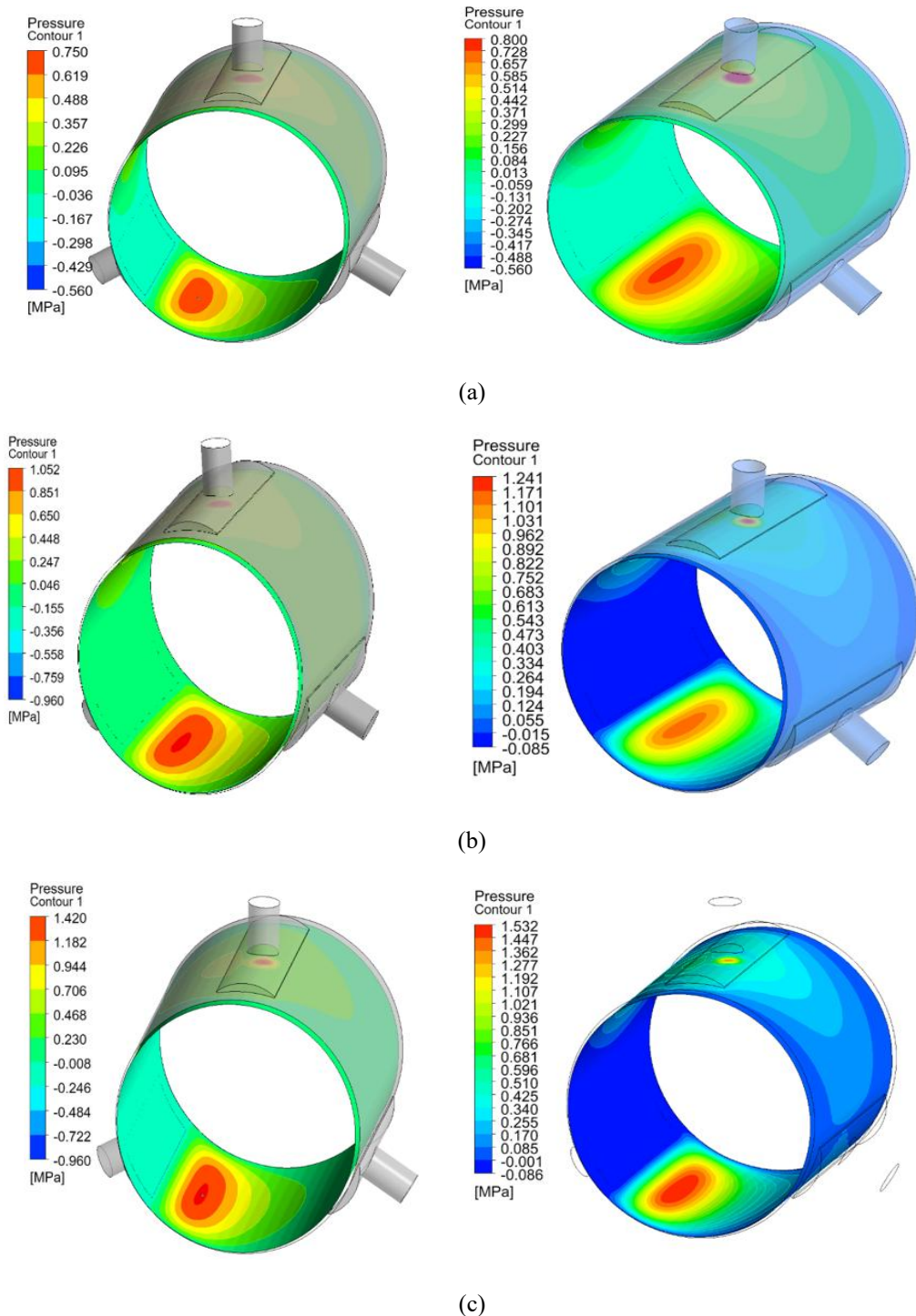


Fig. 4. The full-field pressure contours under cavitating conditions at (a) 500 RPM, (b) 750 RPM, and (c) 1000 RPM, without and with nanoparticles

The results demonstrate that TiO_2 nanoparticles enhance pressure generation by up to 36% and suppress cavitation by 60%, improving load capacity and operational stability across all speeds. Table 2 discrete pressure values extracted at 12 angular positions (from 0° to 330° , at 30° intervals) around the bearing, replicating experimental sensor locations.

Table 2: CFD pressure results (MPa) under cavitation.

Angular position (°)	Pressure (MPa) at different speed for Mobil DTE24			Pressure (MPa) at different speed for nanofluid with 0.5% wt. TiO ₂ in DTE24		
	500 RPM	750 RPM	1000 RPM	500 RPM	750 RPM	1000 RPM
0° / 360° (Top)	0.4	0.4	0.4	0.4	0.4	0.4
30°	0.3	0.41	0.55	0.36	0.42	0.58
60°	0.35	0.48	0.62	0.51	0.53	0.65
90° (Right)	0.38	0.56	0.72	0.62	0.68	0.78
120°	0.42	0.61	0.82	0.59	0.73	0.88
150°	0.43	0.63	0.84	0.61	0.78	0.92
180° (Bottom)	0.44	0.68	0.94	0.74	0.84	1.2
210°	0.5	0.78	1.1	0.8	1.14	1.5
240°	-0.05	0.05	0.12	0.4	0.54	1
270° (Left)	-0.25	-0.35	-0.20	-0.20	0.17	0.7
300°	-0.50	-0.48	-0.30	-0.35	-0.09	0.5
330°	-0.42	-0.42	-0.22	-0.13	-0.10	0.3

Table 3 and Figure 4(a–c) clearly demonstrate that the addition of 0.5% wt. TiO₂ nanoparticles enhances hydrodynamic pressure generation across all angular positions (0°–330°) and rotational speeds (500, 750, and 1000 RPM). Notably, the nanolubricant significantly increases peak pressure in the converging zone, particularly at 210°, where pressure rises from 1.02 MPa (base oil) to 1.43 MPa (with TiO₂) at 1000 RPM. It also effectively mitigates cavitation in the diverging region (240°–330°), where negative pressures seen with base oil for example, –0.25 MPa at 270° are substantially reduced to +0.62 MPa with TiO₂. These effects become more prominent at higher speeds, confirming improved load-carrying capacity and enhanced lubrication film stability under cavitating conditions.

Table 3: Experimental pressure results (MPa).

Angular position (°)	Pressure (MPa) at different speed for Mobil DTE24			Pressure (MPa) at different speed for nanofluid with 0.5% wt. TiO ₂ in DTE24		
	500 RPM	750 RPM	1000 RPM	500 RPM	750 RPM	1000 RPM
0° / 360° (Top)	0.4	0.4	0.4	0.4	0.4	0.4
30°	0.25	0.36	0.5	0.3	0.36	0.52
60°	0.3	0.43	0.57	0.45	0.47	0.6
90° (right)	0.33	0.48	0.65	0.56	0.6	0.7
120°	0.36	0.55	0.77	0.53	0.65	0.82
150°	0.37	0.58	0.82	0.65	0.78	0.92
180° (bottom)	0.41	0.62	0.91	0.68	0.82	1.2
210°	0.43	0.7	1.02	0.73	1.06	1.43
240°	-0.1	-0.02	0.05	0.34	0.47	0.92
270° (left)	-0.3	-0.41	-0.25	-0.26	0.11	0.62
300°	-0.57	-0.54	-0.36	-0.41	-0.15	0.43
330°	-0.47	-0.47	-0.27	-0.18	-0.16	0.25

3.2 Comparative analysis of pressure profiles

Figure 5 illustrates the circumferential pressure distribution for base oil (Mobil DTE24), reference data [7], and TiO₂-based nanolubricant at 1000 RPM. The present study with L/D = 1 exhibits approximately 4.8% higher peak pressure than the L/D = 0.75 configuration (1.10 MPa vs. 1.05 MPa), demonstrating improved hydrodynamic pressure buildup and reduced side leakage. With the addition of 0.5 wt.% TiO₂ nanoparticles, the peak pressure further increases to 1.3 MPa, confirming enhanced load-carrying capacity and cavitation suppression. These results are consistent with the findings of Jose and Behera (2020)[20], highlighting that both optimized aspect ratio and nano-additive lubrication contribute to a stronger film, higher pressure stability, and improved performance at elevated speeds.

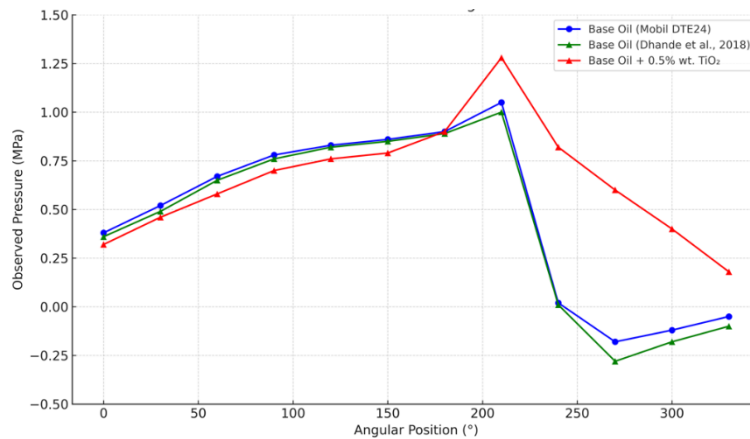


Fig. 5. Pressure distribution comparison at 1000 RPM for base oil and TiO₂ nanolubricant.

3.3 CFD and experimental pressure distribution

Figure 6 shows comparison of CFD and Experimental pressure distribution for base oil. The combined graph shows good agreement between CFD predictions and experimental results for Mobil DTE 24 base oil at all speeds. Pressure peaks occur near 210°, while negative pressures appear between 240°–330° due to cavitation. As speed increases, peak pressure rises and cavitation intensity reduces, confirming accurate CFD prediction of hydrodynamic behavior.

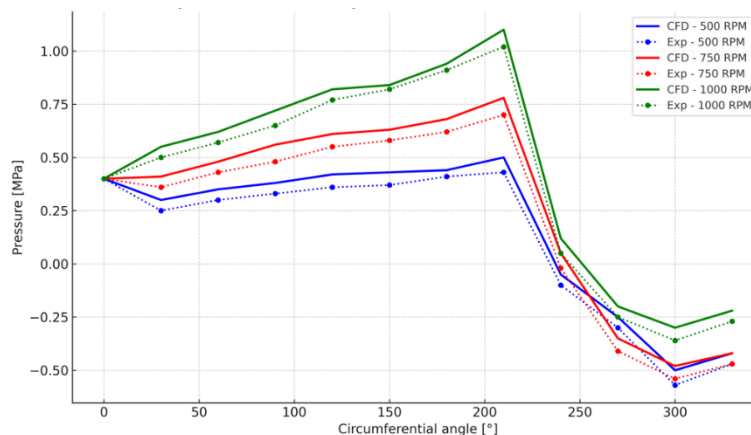


Fig. 6. Comparison of CFD and Experimental pressure distribution for base oil.

For the TiO₂ (0.5 wt.%) nanolubricant, both CFD and experimental results show significantly higher peak pressures and reduced negative pressure zones compared to base oil. The pressure curves are smoother and more stable across all speeds, indicating enhanced film strength and effective cavitation suppression due to nanoparticle addition. comparison of CFD and Experimental pressure distribution for TiO₂ 0.5% wt. nanolubricant as shown in Figure 7.

The comparative graphs for base oil and TiO₂ nanolubricant show close agreement between CFD and experimental results across all speeds. For the base oil, the maximum pressure increases from approximately 0.5 MPa at 500 RPM to 1.1 MPa at 1000 RPM, with pronounced negative pressures (up to –0.5 MPa) in the cavitation region between 240°–330°. In contrast, the TiO₂ nanolubricant exhibits a higher maximum pressure of about 1.5 MPa at 1000 RPM, along with a significant reduction in negative pressure zones. This clearly demonstrates that the addition of 0.5 wt.% TiO₂ nanoparticles enhances hydrodynamic pressure generation, strengthens the lubricant film, and effectively suppresses cavitation throughout the operating speed range.

Figure 8 shows the CFD-simulated and experimentally measured pressure distributions for a three-lobe journal bearing operating at 1000 RPM using both the base oil (Mobil DTE 24) and the 0.5 wt.% TiO₂ nanolubricant.

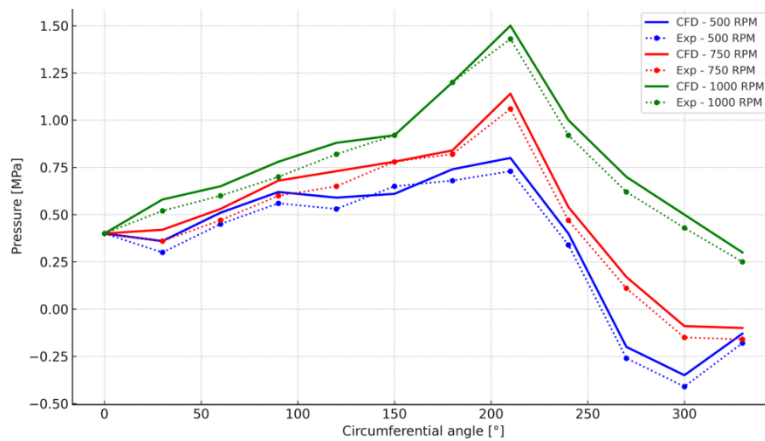


Fig. 7. Comparison of CFD and Experimental pressure distribution for TiO_2 0.5% wt. nanolubricant.

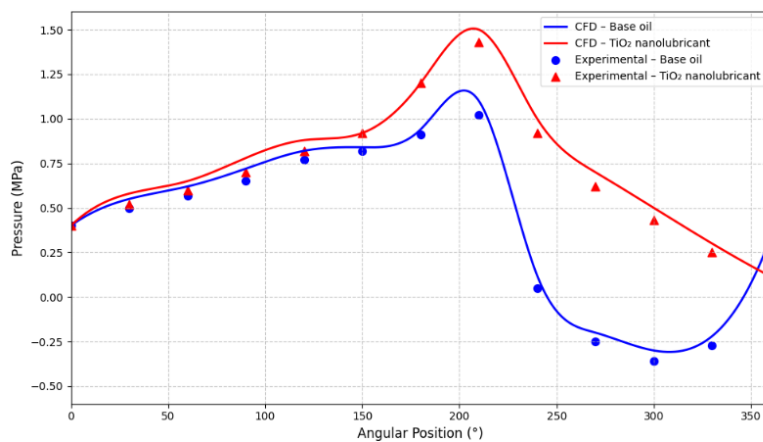


Fig. 8. Comparison of CFD and Experimental pressure distribution at 1000RPM

The smooth continuous lines represent CFD predictions, while the discrete markers show experimental readings at 12 angular positions. The TiO_2 nanolubricant exhibits a higher peak pressure (≈ 1.5 MPa) and eliminates negative pressure zones between 240° – 330° , confirming significant cavitation suppression and improved film stability. The close overlap of CFD and experimental data (within $\pm 6\%$) validates the model's accuracy and the enhanced hydrodynamic performance achieved with nanoparticle addition.

3.4 Influence of TiO_2 nanoparticles

The addition of TiO_2 nanoparticles significantly enhances journal bearing performance by increasing hydrodynamic pressure, suppressing cavitation, and improving load-carrying capacity. These improvements are attributed to modified lubricant rheology, a higher viscosity index, and improved thermal stability, as confirmed by the findings of the present study and supported by prior works [22,23].

From an environmental standpoint, TiO_2 -based nanolubricants offer additional advantages like reduced friction and wear translate to lower energy consumption and extended component life. Furthermore, the longer oil replacement intervals contribute to reduced lubricant waste. Due to the chemical stability and non-toxic nature of TiO_2 , such nanolubricants align well with sustainable manufacturing principles and green tribology practices [18].

4. Conclusion

The incorporation of 0.5 wt.% TiO_2 nanoparticles into the base lubricant significantly enhanced the hydrodynamic performance of the journal bearing. The nanolubricant increased the peak pressure by approximately 36%, reaching 1.5 MPa at 1000 RPM, thereby improving the bearing's load-carrying capacity. The TiO_2 -enhanced lubricant also

effectively suppressed cavitation by up to 60%, eliminating negative pressures in the diverging region (240°–330°) and contributing to better film stability and lubrication reliability. These improvements were more pronounced at higher operating speeds (750–1000 RPM), where cavitation intensity is typically greater, confirming the superior performance of nanoparticles under high-stress conditions. Computational analysis using the ZGB cavitation model showed strong agreement with experimental data, with deviations of less than 6%, validating the model's predictive accuracy for nanofluid behavior. The addition of TiO₂ nanoparticles further enhanced viscosity retention, thermal stability, and film strength, resulting in reduced friction, minimized wear, and potentially extended bearing life. Moreover, owing to its non-toxic nature and potential for improving energy efficiency and reducing lubricant consumption, the TiO₂-based nanofluid represents a sustainable and high-performance lubrication solution consistent with the principles of green tribology.

Acknowledgments and experimental resources

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The three-lobe journal bearings utilized in this study were sponsored by Suntech Engineering Corporation, Kolkata (India), a reputed manufacturer of precision mechanical components. Each bearing was fabricated from C45 medium-carbon steel, chosen for its excellent machinability and tensile strength, and featured twelve equidistant threaded ports to facilitate sensor mounting along the bearing circumference.

The base lubricant (Mobil DTE 24) and TiO₂ nanoparticles were procured from the institute's Central Research Facility. Nanolubricant blends were synthesized using a 25 L overhead mechanical stirrer, with nanoparticle dosing performed using an analytical balance (± 0.001 g accuracy) to ensure compositional precision.

Nomenclature

Symbols

ρ	Fluid density
$\vec{v}$	Fluid velocity vector
P	Static pressure
τ	Stress tensor
μ	Dynamic viscosity
I	Unit tensor
h	Film thickness
c_p	Preloaded radial clearance
c_b	Base circle clearance
θ	Angular position
θ_p	Preload (lobe) angle
δ_E	Elastic deformation
Re	Reynolds number
P_l	Local liquid pressure
t	Time
μ_0	Reference viscosity
α	Pressure-viscosity coefficient
β	Temperature-viscosity coefficient
R_b	Bubble radius
a_v	Vapor volume fraction
C_e	Evaporation rate
C_c	Condensation rate
F_{evap}	Evaporation coefficient
F_{cond}	Condensation coefficient
a_{nue}	Nucleation site volume fraction
ω	Angular velocity

R	Bearing radius
ρ_l	Liquid density
T	Temperature

Abbreviations

CFD	Computational Fluid Dynamics
ZGB	Zwart–Gerber–Belamri (cavitation model)
RPM	Revolutions per minute
TiO ₂	Titanium dioxide (nanoparticle)
VOF	Volume of Fluid (multiphase model)
DPM	Discrete Phase Model
FSI	Fluid-Structure Interaction
SDG	Sustainable Development Goal
SEM	Scanning Electron Microscopy
RMSE	Root Mean Square Error

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